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Use of parabolic trough solar collectors for solar refrigeration and air-conditioning applications

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ABSTRACT

The increasing energy demand for air-conditioning in most industrialized countries, as well as refrigeration requirements in the food processing field and the conservation of pharmaceutical products, is leading to a growing interest in solar cooling systems. So far, the more commonly systems used are single-effect water/lithium bromide absorption chillers powered by flat-plate or evacuated tube collectors operating with COP of about 0.5–0.8 and driving temperatures of 75–95 °C. In general terms, performance of thermally driven cooling systems increases to about 1.1–1.4 using double-effect cycles fed by higher temperature sources (140–180 °C). If solar energy is to be used, concentrating technologies must be considered. Although some experiences on the integration of parabolic trough collectors (PTC) and Fresnel lenses in cooling installations can be found in the literature, the quantity is far to be comparable to that of low temperature collectors. Some manufacturers have undertaken the development of modular, small, lightweight and low cost parabolic collectors, compatible for installation on the roofs of the buildings aiming to overcome some of the current technology drawbacks as costs and modularity. After a comprehensive literature review, this work summarises the existing experiences and realizations on applications of PTC in solar cooling systems as well as present a survey of the new collectors with potential application in feeding double effect absorption chillers. In addition to this, it is evaluated its use as an occasional alternative to other solar thermal collectors in air conditioning applications by dynamical simulation. Results for the case studies developed in this work show that PTC present similar levelized costs of energy for cooling than flat plate collector (FPC) and lower than evacuated tube collectors (ETC) and compound parabolic collectors (CPC).

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1. Introduction

1.1. Overall framework

The energy demand associated with air-conditioning in most industrialized countries has been increasing noticeably in recent

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years, causing peaks in electricity consumption during warm weather periods. This situation is provoked by improved living standards and building occupant comfort demands and architectural characteristics and trends, such as an increasing ratio of transparent to opaque areas in the building envelope [1]. The above, along with refrigeration requirements in the food processing field and the conservation of pharmaceutical products in developing countries, are leading the interest in air-conditioning and refrigeration systems powered by renewable energies, especially solar thermal, which work efficiently, and in certain cases, approach competitiveness with conventional cooling systems.

Solar thermal systems, in addition to the well-known advantages of renewable resources (environmentally-friendly, naturally replenished, distributed,...), are very suitable for air-conditioning and refrigeration demands, because solar radiation availability and cooling requirements usually coincide seasonally and geographically [2,3]. Solar air-conditioning and refrigeration facilities can also be easily combined with space heating and hot-water applications, increasing the yearly solar fraction of buildings.

In spite of the tremendous research effort made in theoretical analysis and experimental projects since the 70s, and the enormous interest related to solar air-conditioning and refrigeration systems, their commercial implementation is still at a very early stage, due mainly to the high costs associated with these systems and the clear market supremacy of conventional compression chillers. Other obstacles to their large-scale application are the shortage of small power equipment and the lack of practical experience and acquaintance among architects, builders and planners with their design, control and operation [4].

In the Solar Heating and Cooling Technology Roadmap published by International Solar Energy in 2012 [5], it is mentioned a study by Mugnier and Jakob [6] estimating in 750 the number of installed solar cooling systems worldwide in 2011, including small capacity (<20 kW) plants. The IEA roadmap also mentions recent developments of big plants, as that of the United World College in Singapore, completed in 2011, with a cooling capacity of 1470 kW and a collector field of 3900 m^2 executed on an energy services company (ESCO) model. Also according to the findings of the study, there is a big potential market for residential applications in Central Europe and in dry and sunny climates areas (Middle East, Australia, Mediterranean islands) although it is constrained by the scarcity of technology options and the difficulties to profit the economy of scale, as well as its dependence on the overall construction sector trends. For example, the case of the Spanish market fall, where the economic crisis has caused a sharp slowdown in growth in the number of projects for residential installations after to lead in 2007 the worldwide market of small size solar cooling systems.

Regarding the type, use and location of the systems, according to the Task 38 of the International Energy Agency survey dated November 2009 [7] the prevailing, 74% for large scale and 90% for small scale (<20 kW), solar cooling installations are those based on thermally driven closed sorption cycles [8,9] especially LiBr–H_2O single-effect absorption chillers [10] fed by flat-plate and evacuated tube collectors, 46% and 40%, respectively, for large systems. Most of the facilities at 2009 were installed in Europe, having Spain, Germany and Italy more than the 60% of the worldwide solar cooling power at that moment. In regard to the use, air conditioning was the main application because two thirds of small scale systems were devoted to offices and private buildings and the half of the large scale systems were used in offices. The geographical distribution of facilities is expected to be spread to other areas as Asia [11] because the present limitations for projects financing in Europe.

Apart from the studies by the International Energy Agency (IEA) in the Solar Heating and Cooling (SHC) program initiated in Task 25Solar-Assisted Air-Conditioning of Buildings [12], ended in 2004 and followed from 2006 to 2010 by Task 38, Solar Air-Conditioning and Refrigeration [13] and Task 48 (2011–2015) Quality Assurance and Support Measures for Solar Cooling [14], it can be mentioned also UE funded international research, development and technology transfer initiatives as the CLIMASOL project [15] where a complete survey about all the different techniques related to solar air conditioning, useful information and advises as well as in depth description of more than 50 working installations in different countries of the projects partners (Germany, France, Spain, Italy and Greece) were done and continued by the Solar Air Conditioning in Europe SACE project, financed by the European Commission, in which five countries participated analysing about 54 solar air-conditioning facilities [4,16], SOLAIR (Solar Air-Conditioning) aimed to promote and to strengthen the use of solar air-conditioning systems on residential and commercial applications [17], SOLARCOMBI+ implemented to achieve a better market of small scale solar cooling systems in combination with traditional domestic solar hot water and space heating [18]. HIGH COMBI (High solar fraction heating and cooling system with combination of innovative components and methods) [19] and MEDISCO, MEDiterranean food and agro Industry applications of Solar COoling technologies [20]. In Spain, although experimental and demonstration solar air-conditioning and refrigeration installations are relatively recent [21], apart from the participation of Spanish institutions and companies in the above mentioned projects and programs, there are good references on on-going works in simulation and design tools [22–32] and experimental evaluation of systems [33–40] as well as reference national programs as ARFRISOL [41].

1.2. High temperature source solar cooling technology

The solar cooling systems can be divided [1,8,42] into: *Closed-cycle thermal systems* divided in *absorption* and *adsorption* and are rated by their coefficient of performance (COP), which is the ratio of cooling energy produced to thermal energy required; *Open-cycle thermal systems* based on a combination of air dehumidification by a desiccant that may be liquid or solid, with evaporative cooling; and *Mechanical systems* use of a solar-powered Rankine-cycle engine connected to a conventional air conditioning system. These can be operated by any type of solar collector, even with PTC [43].

As above mentioned, most of worldwide studies and experiments are based to date on the use of stationary flat-plate collectors (FPC) [29], evacuated tube collectors (ETC), and, to a lesser extent, compound parabolic concentrators (CPC), as the solar heat source in an appropriate operating temperature range (below 100 °C), mainly to feed a single-effect absorption chiller, but also an adsorption chiller or desiccant cooling system [1,4,13,16,44].

In addition to this, the recent developments in gas-fired double-effect absorption LiBr–H_2O chillers, have made available in the market systems with COP 1.1–1.2 that may high temperature solar-powered sources [45] representing, together to the existing experiences with NH_3–H_2O absorption chillers, a promising alternative provided investment and maintenance costs of solar concentrators are reasonable. Linear concentrators, either parabolic trough [46–50] and Fresnel [25,33,51–53] are well suited for this function.

The double-effect absorption cycle permits to take advantage of the higher driving temperature presenting a higher COP, around twice of single-effect cycles [54]. There are two basic configurations for the double-effect cycle depending upon the distribution of the solution through the components, *series* and *parallel* flow.

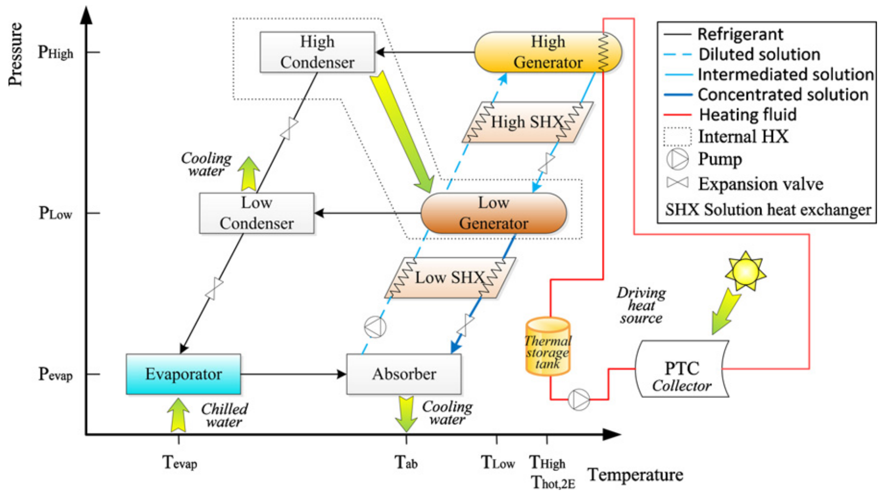

Fig. 1. Diagram of double-effect (2E) LiBr–H_2O absorption chiller (parallel flow). Hot water driving with PTC.

The double-effect absorption cycle (Fig. 1) is characterised by the existence of two desorbers, the high-pressure generator (High-G) and the low-pressure generator (Low-G). PTC collectors provide external heat and vapour is generated in the High-G. The vapour flows to the Low-G, where it changes phase by rejecting heat at sufficiently high temperature that it can be used to separate vapour from the solution flowing through the Low-G. A schematic representation of a parallel flow double-effect LiBr–H_2O absorption cycle is shown in Fig. 1. In the present configuration, the diluted solution leaving the absorber is split into two circuits flowing in parallel. One part of the solution flows to the High-G by passing through the High solution heat exchanger (High-SHX), and the other one goes into the Low-G after passing through the Low solution heat exchanger (Low-SHX). Before entering the absorber, the concentrated solution passes through a Low-SHX where an exchange of sensible heat takes place. The concentrated solution reduces its temperature while the diluted solution gets warmer. The vapour generated in the Low-G is driven directly to the condenser. Finally, the refrigerant separated in the High-G (and condensed in the Low-G) restores its thermodynamic conditions before entering the evaporator. Because optimised use of heat increasing the number of effects.

2. Parabolic-trough collectors for process heat and solar cooling applications

PTCs are parabolic concentrating systems that focus the direct solar radiation parallel to the collector axis onto a focal line (see Fig. 2). A receiver pipe is installed in this focal line with a heat transfer fluid flowing inside it that absorbs concentrated solar energy from the pipe walls and raises its enthalpy. The collector is provided with a one-axis solar tracking system to ensure that the solar beam falls parallel to its axis.

The tracking mode may be of 1-axis or 2-axis. The solar energy received on the collector surface on the 2-axis moving is greater [55,56] due to the lower angle of incidence. However the most of the PTC are 1-axis solar tracking, due to the lower cost and ease of installation. Concerning the orientation, when the PTC axis orientation is N–S (E–W tracking system) the annual incident energy received on the collector surface ($G_{t,\ \mathrm{annual}}$) is greater, which results in a higher annual collector efficiency and smaller auxiliary energy requirement [57].

PTC solar technology is the best proven and lowest-cost large-scale solar power technology available today, mainly due to, in a

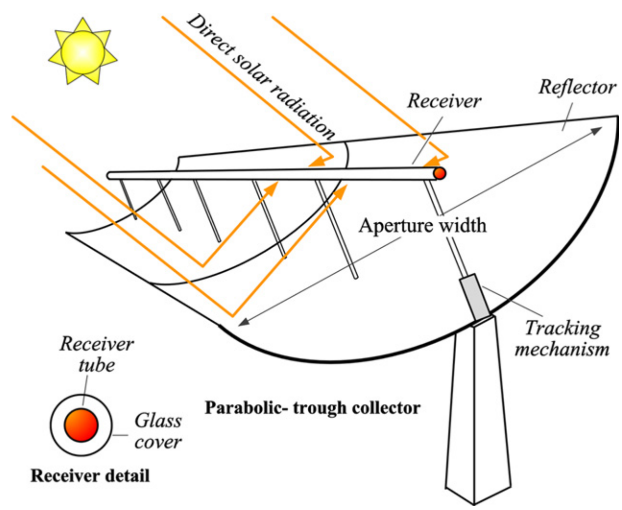

Fig. 2. Schematic of a parabolic-trough collector.

first instance, the accumulated experience of the nine large commercial solar power plants SEGS (solar electric generating systems) with LS-1, LS-2 and LS-3 PTC, developed by *Luz International,* (later acquired by *Siemens-Solel Solar Systems*) in the California Mojave Desert (USA) [58]. PTC reliability has been further confirmed thanks to other models like the *Eurotrough* [59] and *Senertrough* [60] as well as other very recent commercial developments used in the majority of the utility level CSP plants in operation or under construction all over the world. Technological feasibility of direct steam generation (DSG), an outstanding breakthrough in PTC, has already been also demonstrated [61].

PTCs used in solar thermal power plants are very reliable and work at temperatures of around 400 °C and big apertures. However, other eventual applications, such as industrial process heat and heat driven solar cooling, work at maximum temperatures of up to 180 °C and have strong solar field space constraints, because factories and commercial buildings are usually located in places where land is expensive, so rooftop installation should be a real possibility.

In order to ease these applications, some companies have undertaken specific efforts in the development of modular, small, lightweight and low cost PTC. These companies are: *Industrial Solar Technology* (*IST*) [62] (later acquired by *Abengoa Solar*), *Sopogy* [63,64], *Cogenra solar* and *Solargenix energy* in USA;

Table 1
Overview of the main characteristics of small PTC designs in the market.

Manufacturer	Model	Type	Aperture width [m]	Weight [kg/m^2]	Conc. ratio, C [-]	Tracking mode	References
Abengoa Solar	PT-1	PTC	2.3	n/a	14	1-axis	[48,75,76]
	RMT	PTC	1.1	7.7	14	1-axis	[48,62]
Absolicon Solar Concentrator	Absolicon MT 10	PTC	1.1	28.9	n/a	1-axis	[67,77]
	Absolicon T 10	PTC	1.1	28.9	n/a	1-axis	[67,77]
	Absolicon X10 PVT	PTC-PVT	1.1	28.9	n/a	1-axis	[66,67,77]
Cogenra Solar	SunDeck PVT	PLFR-PVT	1.4	49.4	n/a	1-axis	[64,78]
Composites y Sol	CAPSOL	PTC	1.0	n/a	18	1-axis	[68]
DezhouMingnuo New Energy	PT-3E	PTC	3,0	n/a	14	1-axis	[79]
Dr. Vetter	IT. collect	PTC	0.5	14.5	n/a	1-axis	[67,80–82]
Huayuan New Energy Project	HY-TroughII20-2	PTC	2.0	n/a	13	1-axis	[83]
	HY-TroughII30-2	PTC	3.0	n/a	19	1-axis	[83]
	HY-TroughIII20-2	PTC	2.0	n/a	9	1-axis	[83]
	HY-TroughIII30-2	PTC	3.0	n/a	14	1-axis	[83]
IMK	CSP-trough	PTC	2.0	n/a	n/a	1-axis	[84]
Koluacik Research & Development	SPT-0312	PTC	1.2	47.2	8	2-axis	[85]
	SPT-0324	PTC	2.4	33.8	15	2-axis	[85]
	SPT-0424	PTC	2.4	29.2	15	2-axis	[85]
	SPT-0524	PTC	2.4	32.3	15	2-axis	[85]
	SPT-0536	PTC	3.6	27.1	23	2-axis	[85]
NEP Solar	Polytrough 1200	PTC	1.2	25.3	14	1-axis	[48,69,82,86]
	Polytrough 1800	PTC	1.8	17.9	17	1-axis	[82,86]
SMIRRO	Smirro 300	PTC	1.1	14.6	10	1-axis	[81,87]
Solargenix Energy Headquarter	Power Roof	PTC	3.6	n/a	27	1-axis	[88]
Solarlite	SL 2300	PTC	2.3	2.0	n/a	1-axis	[82,89]
Solitem	PTC 1100	PTC	1.1	14.5	n/a	1-axis	[90]
	PTC 1800	PTC	1.8	14.2	15	1-axis	[46,48,69,82]
	PTC 3000	PTC	3.0	14.0	n/a	1-axis	[90]
Soltigua	PTMx	PTC	2.4	n/a	n/a	1-axis	[81,82,91]
Sopogy	SopoHelios	PTC	2.1	10.4	21	1-axis	[92]
	SopoNova	PTC	1.7	13.1	21	1-axis	[63,64,82]
	SopoTitan	PTC	3.0	n/a	n/a	1-axis	[92]
Thermax	SolPac P60	PTC	n/a	n/a	n/a	1-axis	[82,93]
Trivellienergia	SolarWing Evolution	PTC	1.3	18.0	n/a	1-axis	[82,94]
Hitachi Plant Technologies	Prototype	PTC	n/a	n/a	n/a	1-axis	[14]

Solitem[46] *and Koluacik* in Turkey; *NEP Solar* [65] in Australia; *Absolicon Solar Concentrator* [66] in Sweden, *Soliqua* and *Trivelli energia* in Italy; *Dr. Vetter* [67], *Smirro* and *Solarlite* in Germany; *Huayuan new energy project* and *Dezhou Mingnuo New Energy* in China; *IMK* in Austria; *Thermax* in India; and *Composites y Sol* in Spain [68] market this type of collector. Reference to other prototypes as *PARASOL* [69] in Austria; *PTC-1000* [70] and *Prototype-Mithras* in Germany; *CHAPS* [69] in Australia; *Prototypes* [71,72] in Mexico; *Hitachi Plant Technologies* [14] in Japan; and *SALTO PTC* [73,74] in Italy.

According the study carried out in this work, there are over 30 models of small PTC (aperture width lesser than 4 m) that could be used in solar cooling systems. Table 1 summarizes the characteristics of the collectors. Of these, only few have been certified by present standards either by Solar Keymark (SK) [67] or by SRCC [64].

These developments allow have for a larger number of options of high temperature sources for double-effect absorption chillers, fulfilling at the same time basic requirements on costs and operation.

3. Review of cooling systems with PTC

Air-conditioning and refrigeration facilities driven by a PTC solar field are still in frequent despite simulations studies confirms the functional feasibility of this technique in weather favourable regions [95,96]. Several experimental and test facilities using this technology have appeared in the literature. The first two references in the literature go back to 1957. One of them is a prototype designed and erected at the *Laboratoire de l'Energie Solaire* in Montlouis (France), where a 1.5-m^2 east–west-oriented PTC was connected to an intermittent NH_3–H_2O absorption refrigeration cycle. The coolant was heated directly inside the solar collector, producing 6 kg of ice per day [97]. The other facility erected in 1957 was an experimental plant at the *University of Florida* (Florida, USA). This facility had a 4.46-m^2 PTC collector which used cottonseed oil at around 290 °C as the working fluid, and supplied thermal energy to an absorption chiller for air-conditioning [98].

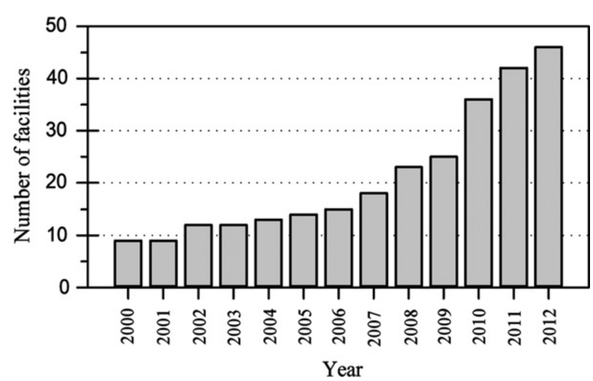

Fig. 3. Yearly evolution of PTC facilities connected to solar cooling systems.

A parabolic-trough collector field was installed on the roof of a building near Kuwait City (Kuwait). The solar field aperture was 63.18 m^2 and it was used to supply thermal energy to a commercial LiBr–H_2O single-effect absorption chiller, the 10.55-kW (3-ton) *ARKLA Solaire* Model 501-WF, for air-conditioning. The heat transfer fluid employed was pressurized water and the thermal storage subsystem consisted of an 8000-l water storage tank. The unit required a water temperature of 98.9 °C in the

Table 2
PTC facilities connected to solar cooling systems. (Enhanced of [48,108–111]).

Location|application	A_{spec} [m^2/kW] or (A_c [m^2])	V_{sto}/A_c [m]	Fluid	Collector producer	Year	Chiller type|fluid pairs|producer|cooling capacity [kW]	Propose	References
Florida, USA| *University of Florida*	(4.5 m^2)	n/a	Thermal oil	Prototype	1957	1E|NH_3–H_2O|prototype|n/a	SC	[98]
Montlouis, France|*Laboratoire de l'EnergieSolaire*	(1.5 m^2)	n/a	n/a	Prototype	1957	1E|NH_3–H_2O|prototype|n/a	Ice	[97]
Dacca, East Pakistan|*East Pakistan University*	n/a	n/a	n/a	Prototype	1964	1E|NH_3–H_2O|prototype|n/a	SC	[112]
Yuma (AZ), USA| *U.S. Army Yuma Proving Ground*	2.21	n/a	Water	Hexel	1979	2E|LiBr–H_2O|n/a|563	SHC, DHW	[113]
Kuwait City, Kuwait| *Building*	5.99	0.13	Water	n/a	1984	1E|LiBr–H_2O|ARKLA|11	SC	[99]
Juzbado (Salamanca), Spain| *Empresa Nacional de Uranio (ENUSA)*	(1080 m^2)	n/a	Heliotermo 2550	n/a	1985	1E|n/a|n/a|n/a	SC	[101]
Kharagpur, India| *Institute of Technology Solar Energy Laboratory*	(1.5 m^2)	n/a	n/a	Prototype	1989	1E|n/a|Himalux|n/a	Food preservation	[100]
Amman, Jordan| *University of Jordan*	(3.8 m^2)	n/a	water	Prototype	1992	1E|LiBr–H_2O|prototype|n/a	SC	[102]
Spain| *Polytechnic University of Madrid*	n/a	n/a	n/a	Prototype	2000	1E|NH_3–H_2O| prototype|2	SC	[103]
Abu Simbel, Egypt| *ISAAC Solar Icemaker*	(12 m^2)	n/a	n/a	Prototype	2002	n/a|n/a|n/a|n/a	Ice	[114]
Ankara, Turkey| *Gazi University*	n/a	n/a	n/a	Prototype	2002	n/a|NH_3–H_2O|prototype|n/a	SC	[115]
Raleigh (NC), USA| *Parker-Lincoln building*	2.67	0.05	Pressur. water	Solargenix Energy	2002	2E|LiBr–H_2O|Broad|176	SHCP, DHW	[88]
Dalaman, Turkey| *IberotelSarigerme Park*	2.57	n/a	Pressur. water	Solitem	2004	2E|LiBr–H_2O|Broad|140	SC, Steam	[46,116,117]
Alanya, Turkey| *Grand Kaptan*	2.40	n/a	Pressur water	Solitem	2005	2E|LiBr–H_2O|Broad|150	SC	[90,117]
Douglas (AZ), USA|*Cochise College Campus*	3.01	0.04	Water and glycol	Abengoa solar	2006	1E|NH_3–H_2O|n/a|210	SC	[118]
Brisbane (QLD), Australia| *Ipswich Hospital*	1.93	0.01	Pressur. water	NEP Solar	2007	2E|LiBr–H_2O| Broad|295	SC	[119]
Newcastle, Australia|*REDI Solar Cooling Demo. Project (CSIRO)*	2.78	n/a	Pressur. water	NEP Solar	2007	1E|LiBr–H_2O|Yazaki|18	SC	[119]
Pittsburgh (PA), USA| *Carnegie Mellon University*	3.25	0.08	Pressur. water	Broad	2007	2E|LiBr–H_2O|Broad|16	SHC	[120]
Antalya, Turkey| *Metro Cash & Carry*	1.69	0.03	Pressur. water	Solitem	2008	2E|LiBr–H_2O| Broad|250	SHC	[90]
Cologne, Germany|*REACt project*	(150 m^2)	n/a	Pressur. water	Solitem	2008	n/a|n/a|n/a|n/a	SHC, Steam	[121]
Gebze, Turkey|*Gebze High technology Institute*	1.25	n/a	Pressur. water	Solitem	2008	2E|LiBr–H_2O|broad|260	SC, DHW	[117,122]
Padstow (NSW), Australia| *SERDF Solar Cooling Demo. Proj.*	0.94	n/a	Pressur. water	NEP Solar	2008	2E|LiBr–H_2O|broad|175	SC	[119]
Tarsus, Turkey|*FritolayPepsico*	3.43	n/a	Pressur. water	Solitem	2008	n/a|n/a| n/a|420	SC, Steam	[90]
Barcelona, Spain|*Office-building* (*Controlmatic Headquarters*)	3.04	n/a	n/a	Siemens	2009	1E|LiBr–H_2O|LG|70	SHC	[123]
Downey (CA), USA|*SoCalGas' Energy Resource Center* (*ERC*)	2.42	n/a	n/a	Sopogy	2009	1E|LiBr–H_2O| Yazaki|35	SC	[124]
Araluen (NT), Australia| *Art gallery*	n/a	n/a	n/a	NEP Solar	2010	n/a|n/a| n/a|n/a	SC	[86]
Casablanca, Morocco| *Moulay Youssef Hospital*	8.31	n/a	Thermal oil	Solitem	2010	1E|NH_3–H_2O| Robur|13	SHC	[90]
Dead sea, Jordan| *Dead Sea hotel*	9.69	n/a	pressur. water	Solitem	2010	1E|NH_3–H_2O|Robur|13	SC	[90]
Firenze, Italy|*Misericordia*	6.35	n/a	Saturated steam	Solitem	2010	1E|NH_3–H_2O|Robur|17	SHC	[90]
Germany|*ITW-Universität Stuttgart*	n/a	n/a	n/a	Smirro	2010	1E|LiBr–H_2O|Invensor|10	SHC	[87]
Härnösand, Sweden|*Hospital*	3.67	n/a	n/a	Absolicon	2010	2E|LiCl–H_2O|ClimateWell|10	SHCP	[77]
Long Island City (NY), USA|*Steinway & Sons*	1.69	n/a	Water and glycol	Abengoa solar	2010	2E|LiBr–H_2O|Broad|317	SC	[125,126]
Marrakech, Morocco| *Politecnico di Milano*	4.57	n/a	Thermal oil	Abengoa solar	2010	1E|NH3–H_2O|Robur|13	SC	[108]
Newcastle, Australia| *Cinema Complex*	1.54	n/a	pressur. water	NEP Solar	2010	2E|LiBr–H_2O|n/a|230	SC	[119]
North republic of Cyprus| *METU campus*	5.82	n/a	pressur. water	Solitem	2010	1E|LiBr–H_2O| Thermax|130	SHCP	[90]
Pavia, Italy| *Office-building*	n/a	n/a	n/a	Trivellienegia	2010	2E|LiBr–H_2O| Broad|70	SHC	[94]
Abu Dhabi, United Arab Emirates| *Masdar city*	3.24	0.03	Thermal oil	Sopogy	2011	2E|LiBr–H_2O| Broad|176	SC	[92]
El Paso (TX), USA| *Fort Bliss*	5.66	n/a	water	Sopogy	2011	1E|LiBr-H_2O|n/a|141	SC	[92]
Gambettola (FC), Italy| *Office-building*	n/a	n/a	n/a	Soltigua	2011	2E|LiBr–H_2O|n/a|n/a	SHC	[91]
Gurgaon (Haryana), India| *Solar Energy Center*	n/a	n/a	n/a	Thermax	2011	3E|absorption|Thermax|100	SHC, Steam	[104]
Seville, Spain|*Centro Tecnológico Campus Palmas Altas*	2.48	n/a	n/a	Abengoa solar	2011	n/a|n/a| n/a|63	SC	[127]
Tuscon (AZ), USA| *Davis-Monthan Air Force Base*	3.23	n/a	water	Sopogy	2011	2E|LiBr-H_2O| n/a|211	SC	[92]
Downey (CA), USA| *SoCalGas' Energy Resource Center (ERC)*	2.01	n/a	n/a	Cogenra	2012	1E|LiBr–H2O|n/a|35	SCP	[78]
Malta|*Arrow Pharm* (*DiGeSPo Project*)	n/a	n/a	Therminol 66	Prototype	2012	Electricalcoolingsystem	Power	[107]
Mohali, India| *Hospital*	3.17	n/a	n/a	Absolicon	2012	n/a|n/a| n/a|32	SHCP	[77]
Phitsanulok, Thailand| *TRESERT Phitsanulok*	8.84	n/a	n/a	Solarlite	2012	n/a|n/a|n/a|105	SHCP	[89]

Table 2 (*continued*)

Location|application	A_{spec} [m^2/kW] or (A_c [m^2])	V_{sto}/A_c [m]	Fluid	Collector producer	Year	Chiller type|fluid pairs|producer|cooling capacity [kW]	Propose	References
Hermosillo, Mexico| *Holcim*	5.82	n/a	water	Sopogy	2013	1E|LiBr-H_2O| n/a|264	SC	[92]
Karak, Jordan|*Trigeneration at Mu'Tah University*	(371 m^2)	n/a	Xceltherm-600	Sopogy	2013	n/a|n/a|n/a|n/a	SHCP	[92]
Maalaea (HI), USA|*Maui Ocean Centera Wailuku*	(226.8 m^2)	n/a	Water	Sopogy	2013	2E|LiBr–H_2O|n/a|n/a	SC	[92]

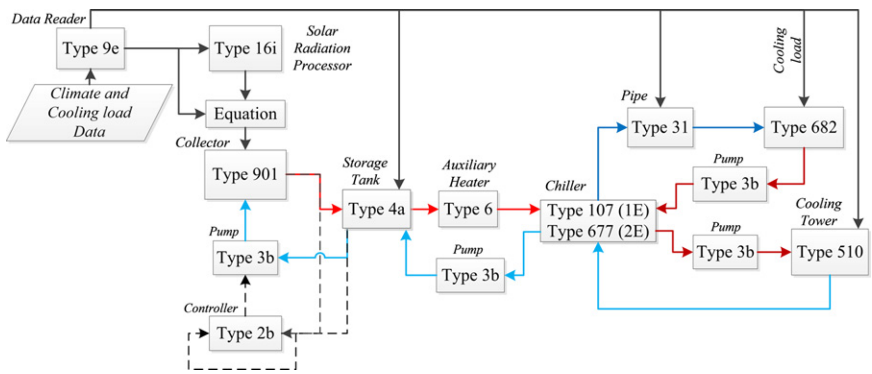

Fig. 4. TRNSYS model of solar cooling installation.

generator for full-capacity cooling with condensing water at 29.5 °C. The average solar field efficiency was 55% [99].

The *Indian Institute of Technology Solar Energy Laboratory* (Kharagpur, India), modified a commercial electric vapour-absorption refrigerator manufactured by *Himalux*, and connected it to a PTC for preservation of perishable food. The solar collector designed and constructed in the same project had a 1.5-m^2 aperture area and 50% average efficiency, and the operating results were good enough to reach 3 °C in the evaporator [100].

A PTC solar field was hooked up to an absorption chiller by the *ENUSA Company*, in Juzbado (Salamanca, Spain). The solar field aperture area was 1080-m^2 and worked with Heliotermo 2550 thermal oil, supplying collector outlet temperatures of around 180 °C. The facility operated satisfactorily from 1985 to 1988 [101].

A continuous solar system with no storage powering a LiBr–H_2O absorption refrigeration cycle was designed and tested at the *University of Jordan* (Amman, Jordan). Both the solar collector and the experimental cooling unit were manufactured locally. The solar field was made up of 3.6-m^2 flat-plate collectors, followed by a 0.15-m^2 PTC to increase inlet generator temperature up to 95 °C. The average coefficient of performance achieved during testing was 0.55 [102].

A 2-kW NH_3–H_2O absorption prototype connected to a PTC for refrigeration in small rural operations was recently developed and tested in Spain. The heat transfer fluid was thermal oil and the collector supplied temperatures up to 150 °C [103].

A system with a 100 kW cooling capacity, has an integrated triple-effect (3*E*) with vapour absorption chiller (VAC), COP_{rated}= 1.7, and PTC has been installed at India in 2011. The collector supplied temperatures up to 210 °C [104].

There are also other facilities of PTC for electricity production by PVT [66] or micro-CSP with organic Rankine-cycle [63], which are used to feed an electrical cooling system. The facilities have also been made with steam jet ejector chiller (SJEC) is a thermo-mechanical chiller for cold generation [105,106] and other installation of *Arrow Pharm* in Malta with electricity generation through Stirling engine connected to a field of PTC, (project DIGESPO) [107].

Fig. 3 shows the yearly evolution of PTC facilities connected to solar cooling systems. In the last 12 years till 37 facilities have been implemented worldwide. An overview of the existing facilities is shown in Table 2.

4. Comparative analysis

The advantages of PTCs over the solar collectors traditionally used in solar air-conditioning and refrigeration facilities are their lower thermal losses and, therefore, higher efficiency at the higher working temperature reached, smaller collecting surface for a given power requirement, and no risk of reaching dangerous stagnation temperatures, since in such cases the control system sends the collector into stow position.

The disadvantages of PTCs are that its solar tracking system increases installation and maintenance costs, the need to clean their components also increases maintenance costs, as PTCs can only use direct solar radiation, their installation is geographically limited, and at very high wind speeds operation must be interrupted and the collectors sent into stow position.

We have devised a method for comparing PTC efficiency to conventional collectors for a given solar-assisted air-conditioning system with a LiBr–H_2O absorption chiller on the basis of the case studies the SACE project [4,16,128] where a compressive analysis of the effects of cooling loads and collector type for low temperature driven chillers has been done. Additional calculations required to provide for variation in stationary collector inclination, and collectors with solar tracking in the analysis has been included and a dynamical approach has been considered by developing a dynamic model in TRNSYS (see Fig. 4). It has also been developed a new Type 901 for calculating the energy produced by different solar collectors, using the design parameters of the efficiency curves and the incidence angle modifiers, taking into account Eqs. (3) and (4).

The calculation method was also extended to study the effect of including thermal storage between the solar field and the cooling system. A certain storage capacity is defined as tank volume per collector area, V_{sto}/A_c. In this case, excess solar energy

at a given hour can be used at a later time when solar gains are insufficient to match the cooling load (affected by a thermal losses coefficient). When the volume of the storage tank increases, the efficiency of collector increases, in relation to that the performance of the systems becomes better, but the temperature of the usable water in tank decreases [129,130].

The key system efficiency parameter employed is the solar fraction, f[-], defined as the fraction of the total load which is covered by the solar energy system. The hourly solar fraction, f_h[-], is found as indicated in Eq. (1).

$$f_h = \frac{E_c\ A_c}{E_{\text{hot}} A_{\text{building}}} = \frac{E_c\ A_c}{(E_{\text{cool}}/COP) A_{\text{building}}} \quad (1)$$

where E_c [W h/m^2] is the mean hourly thermal energy supplied by the solar system per collector area, A_c [m^2], usable by the system; E_{hot} [W h/m^2] is the mean hourly thermal energy demanded by the air-conditioning system per building area, A_{building} [m^2]. E_{hot} can be expressed also as the mean hourly cooling load, E_{cool} [Wh$_{\text{cool}}$/m^2], divided by the COP. f_h=1 if $E_c\ A_c > E_{\text{hot}} A_{\text{building}}$. The mean annual value, f, is found from f_h when $E_{\text{cool}} > 0$.

The specific collector area, A_{spec} [m^2/kW] is the installed solar collector area, per unit of installed cooling capacity, $\dot{Q}_{\text{cool}}$ [kW], Eq. (2).

$$A_{\text{spec}} = \frac{A_c}{\dot{Q}_{\text{cool}}} \quad (2)$$

The files generated in the SACE project [4,16,128] are entered as input data. These files contain annual time series of hourly meteorological data (extracted from a commercial database) and mean hourly cooling load per m^2 of building, E_{cool}. Data files are

Table 3
Data of cases studied. (SACE—Solar Air Conditioning in Europe [128]).

Site	Latitude/ longitude [°]	Office-building		Hotel	
		$A_{building}$ [m^2]	$E_{cool,annual}$ [kW h/(m^2 year)]	$A_{building}$ [m^2]	$E_{cool,annual}$ [k Wh/(m^2 year)]
Madrid, Spain	40.42/−3.70	930.0	28.050	642.4	71.439
Copenhagen, Denmark	55.68/12.57	930.0	13.221	642.4	37.244

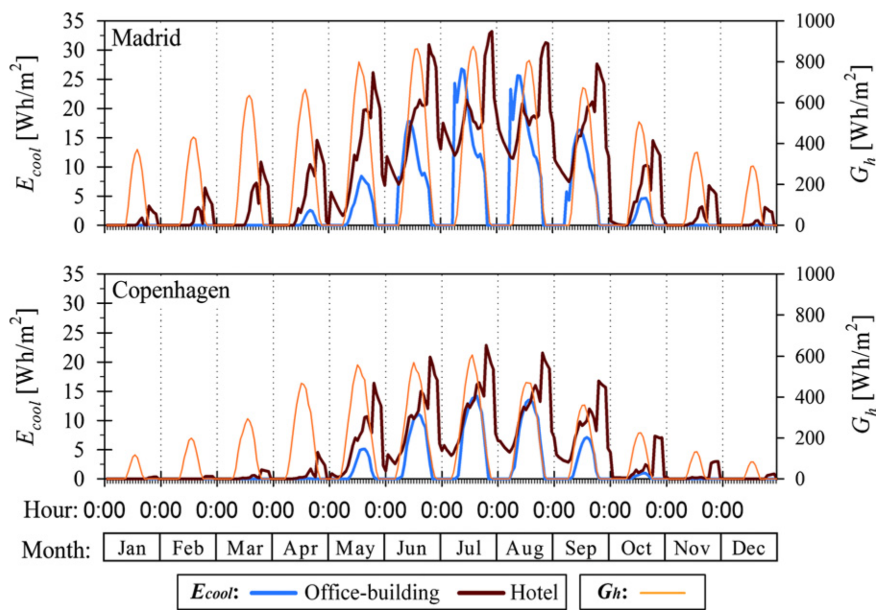

Fig. 5. Yearly profile of average hourly cooling load (E_{cool}) for the office-building and the hotel, and global solar radiation on a horizontal surface (G_h), in Madrid and Copenhagen. Monthly typical days. (SACE—solar air conditioning in Europe [128]).

Table 4
Zero-loss collector efficiency (η_0), coefficients of temperature dependent heat loss coefficient (a_1, a_2), diffuse incident angle modifier ($K_{\theta d}$), direct incident angle modifier coefficients (b_{1L}, b_{2L}, b_{1T}, b_{2T}), and configuration of solar collector models considered.

Type of collector	FPC-1	FPC-2	ETC	CPC	PTC-1	PTC-2	PTC-3
Specific costs	Standard	Standard	Standard	Standard	Standard	Low cost	Standard
Aperture width [m]	2.38 × 1.06	2.38 × 1.06	1.45 × 1.64	2.43 × 1.62	2.3	1.7	1.2
References	[67]	[67]	[67]	[67]	[75]	[64]	[131]
η_0 [-]	0.776	0.791	0.745	0.644	0.6931	0.5897	0.68
a_1 [W/(m^2 K)]	4.14	3.94	2.007	0.749	0.4755	0.9317	0.4
a_2 [W/(m^2 K^2)]	0.0145	0.0122	0.005	0.005	0.003128	0	0.0015
$K_{\theta d}$ [-]	0.840	0.876	0.85 [132]	0.54 [133]	0.070 [134]	0.048 [134]	0.073 [134]
b_{1L} [-]	−1.46E−03	−1.00E−03	−1.37E−03	−1.14E−03	3.18E−04	−3.06E−03	2.20E−04
b_{2L} [-]	−4.50E−07	−3.00E−07	1.10E−05	8.93E−06	−3.99E−05	−7.40E−06	−3.83E−05
b_{1T} [-]	0	0	8.315E−03	9.948E−04	0	0	0
b_{2T} [-]	0	0	−8.896E−05	−9.530E−06	0	0	0
Angle tilted [°]	Lat.-10 [135]	Lat.-10 [135]	Lat.-10 [135]	Lat.-10 [135]	0	0	0
Tracking mode	None	None	None	None	1-axis	1-axis	1-axis
Axis|orientation	|South	|South	|South	|South	N–S|	N–S|	N–S|

available for seven reference locations and three typical reference buildings in each location. This paper discusses data from Madrid and Copenhagen, as they have very different levels of direct solar radiation with respect to global radiation. In each of the two locations, an office-building and a hotel were studied (see Table 3). The annual profile evolution of cooling load is shown in Fig. 5.

Both single-effect (1E) and double-effect (2E) LiBr–H $_2$O, hot water fired absorption chillers of power rated cooling capacity, $\dot{Q}_{cool}=40$ kW, are analyzed. For each type of chiller, the nominal COP_{rated} and driving temperature, T_{hot}, are different. For single-effect ($COP_{rated,\ 1E}=0.7$, $T_{hot,\ 1E}=90$ °C) and for double-effect ($COP_{rated,\ 2E}=1.2$, $T_{hot,\ 2E}=150$ °C).

For the comparative analysis, several representative collectors have been selected taking into account the availability information of performance and costs. The solar collectors models considered are shown in Table 4, and the efficiency curves in Fig. 6.

The mean overall efficiency at every hour of the year to the aperture area, η, it is described by Eq. (3) [136].

$$\eta=\eta_0\,\frac{(K_{\theta b}(\theta)\ G_{b,t}+K_{\theta d}\ G_{d,t})}{G_t}-a_1\frac{(T_m-T_a)}{G_t}-a_2\frac{(T_m-T_a)^2}{G_t} \tag{3}$$

where, η_0 [-] is the zero-loss collector efficiency, a_1[W/(m^2 K)] is the heat loss coefficient, a_2 [W/(m^2 K^2)] is the temperature dependence of the heat loss coefficient, G_t [W/m^2] is the mean hourly global, $G_{b,t}$ [W/m^2] the direct and $G_{d,t}$ [W/m^2] the diffuse, solar radiation on a surface tilted, T_m[K] is the mean heating temperature, T_a [K] the ambient temperature, $K_{\theta d}$ [-] is the diffuse

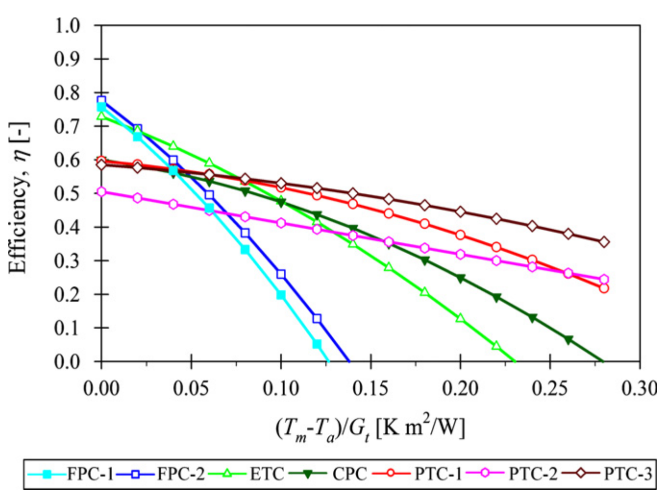

Fig. 6. Efficiency curves of solar collectors considered as a function of difference between mean heating temperature (T_m) and ambient temperature (T_a). [being angle of incidence, $\theta=0°$, and diffuse fraction, $(G_{d,t}/G_t)=0.15$].

incident angle modifier, and $K_{\theta b}(\theta)$ [-] is the direct incident angle modifier for the angle of incidence, θ.

The diffuse incident angle modifier, $K_{\theta d}$, for the PTC with low concentration ratio, C, is assumed equal to $1/C$ [134]. The direct incident angle modifier $K_{\theta b}(\theta)$, without the cosθ, is calculated by the Eq. (4).

$$K_{\theta b}(\theta)=K_{\theta L}(\theta_L)\ K_{\theta T}(\theta_T)=\left(1+\frac{\left(b_{1L}\ \theta_L+b_{2L}\ \theta_L{}^2\right)}{\cos\theta_L}\right)\left(1+\frac{\left(b_{1T}\ \theta_T+b_{2T}\ \theta_T{}^2\right)}{\cos\theta_T}\right) \tag{4}$$

where, $K_{\theta L}$ [-] and $K_{\theta T}$ [-] are the direct modifiers of incident angle longitudinal (θ_L) and transversal (θ_T), $b_{1L,}$ $b_{2L,}$ $b_{1T,}$ b_{2T} [-] are the direct incident angle modifier coefficients.

Annual time series of hourly thermal energy supplied by collectors, E_c, are found using Eq. (5).

$$E_c=\eta\ G_t \tag{5}$$

where, G_t (W h/m^2) is the mean hourly global solar radiation on a surface tilted, received by the collector aperture, calculated by using a Reindl [137] radiation model, with an albedo equal to 0.2, from the annual time series of hourly global, G_h, and diffuse, $G_{d,h}$, horizontal solar radiation contained in the source data files.

For the case of energy storage, the analyzed cases were the office-buildings in Madrid and Copenhagen. The solar collectors selected were the FPC-2 and PTC-1, both connected to single-effect (1E) and double-effect (2E) absorption chillers. Annual fractional thermal energy savings, $f_{sav,therm}$, are found using Eq. (6), takes into account the saved fuel input of the solar system compared to a reference heating system [138].

$$f_{sav,therm}=1-\frac{E_{aux}}{E_{ref}} \tag{6}$$

where E_{aux} [kW h], is the annual auxiliary energy required; E_{ref} [kW h], is the value for the reference, non-solar heating system ($A_c=0$ m^2) without storage ($V_{sto}/A_c=0$ m) and 1E absorption chiller, with which the solar system is compared.

Table 6
Global solar radiation and diffuse fraction received by the collectors.

Location	FPC, ETC, CPC (non-tracking)			PTC (1-axis tracking; axis orientation N–S)		
	Angle tilted [°]	$G_{t,\ annual}$ [kW h/(m^2 year)]	G_{dt}/G_t [-]	Angle tilted [°]	$G_{t,\ annual}$ [k Wh/(m^2 year)]	G_{dt}/G_t [-]
Madrid	30.42	1864.098	0.38	0	2633.197	0.37
Copenhagen	45.68	1165.297	0.49	0	1335.659	0.45

Table 5
Capital costs of solar assisted cooling system considered [$A_{spec}=3$ m^2/kW, $V_{sto}/A_c=0$ m].

Type of collector			FPC-1	FPC-2	ETC	CPC	PTC-1	PTC-2	PTC-3
Investment	Solar collector [€/m^2]		310	430	650	550	310	190	440
	Absorption chiller [€/kW$_{cool}$][141,142]	1E	400	400	400	400	400	400	400
		2E	700	700	700	700	700	700	700
	Others invest. costs [€/kW$_{cool}$]		2000	2000	2000	2000	2000	2000	2000
	Total invest. costs [€/kW$_{cool}$]	*1E*	*3330*	*3690*	*4350*	*4050*	*3330*	*2970*	*3720*
		2E	*3630*	*3990*	*4650*	*4350*	*3630*	*3270*	*4020*
O&M	Solar collector [€/(m^2 year)]		2	2	2.5	3	4	4	4
	Others O&M costs [€/(kW$_{cool}$ year)][140]		17	17	17	17	17	17	17
Fuel	Natural gas [€/(kW h$_{hot}$ year)]		0.05	0.05	0.05	0.05	0.05	0.05	0.05

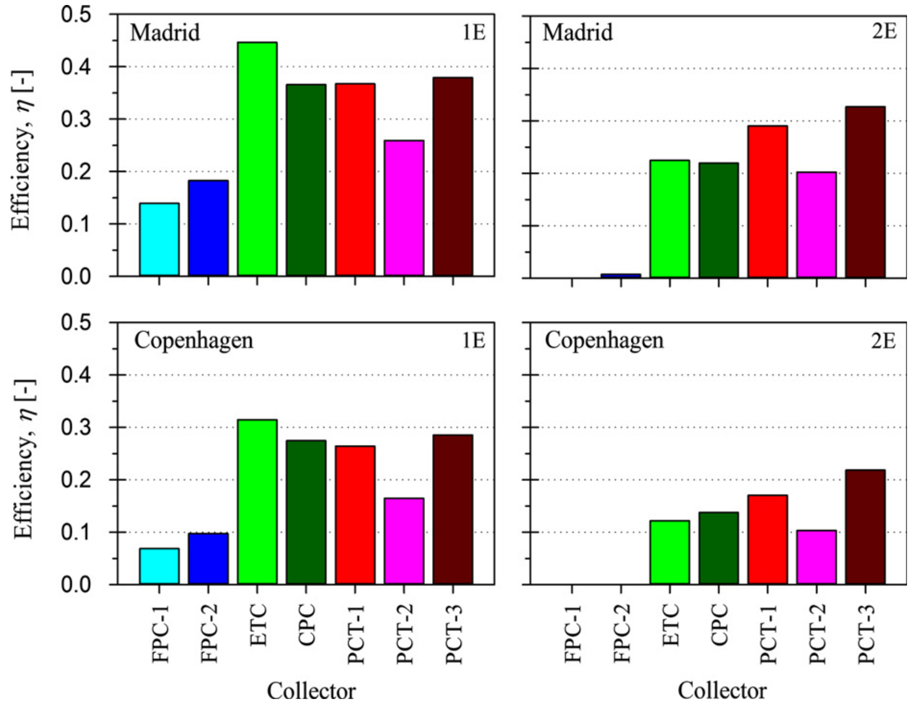

Fig. 7. Annual efficiency of solar collectors considered for the office-building and the hotel in Madrid and Copenhagen, connected to a single-effect (1E) and double-effect (2E) absorption chillers [$A_{spec}=3\ m^2/kW$, $V_{sto}/A_c=0$ m, external conditions are shown in Table 6].

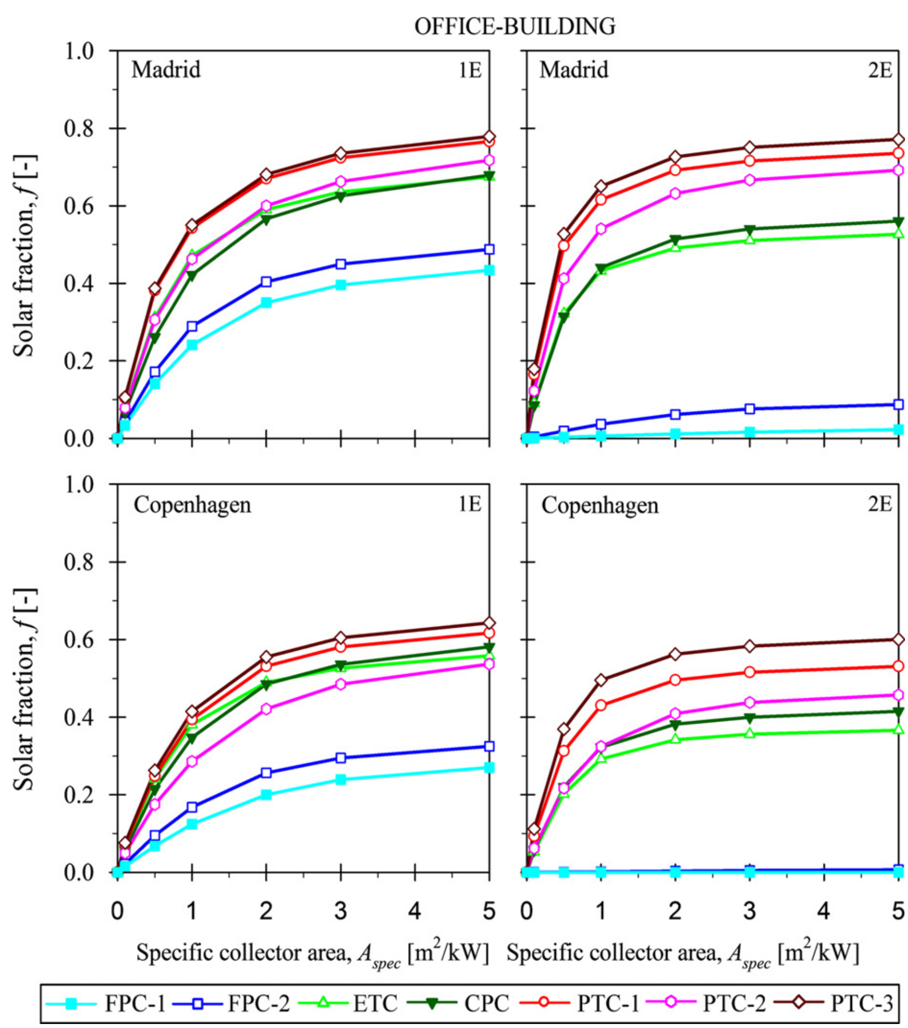

Fig. 8. Collector comparison of the annual solar fraction, for the office-building in Madrid and Copenhagen, connected to single-effect (1E) and double-effect (2E) absorption chillers [$V_{sto}/A_c=0$ m].

Finally, an economic study is performed by calculating the Levelized Cost of Energy cooling, $LCOE_{cool}$ defined as:

$$LCOE_{cool} = \frac{crf\ K_{invest} + K_{O\&M} + K_{fuel}}{E_{cool,annual}\ A_{building}} \tag{7}$$

where K_{invest} is the capital cost of the total investment of the plant [€], $K_{O\&M}$ is the annual cost of operation and maintenance [€/year]; K_{fuel} is the annual fuel cost [€/year] (if you use some other fuel than solar radiation); $E_{cool,annual}$ is the annual net cooling energy produced [kW h_{cool}/(m^2 year)], and crf is the annual amortization factor [-], which is obtained as:

$$crf = k_d \frac{(1+k_d)^n}{(1+k_d)^n - 1} + k_{insurance} \tag{8}$$

where n is the project lifetime in years; k_d is the interest rate debt [-] and $k_{insurance}$ is the annual insurance rate [-].

The considered capital costs for different components of solar assisted cooling system, are show in Table 5. The total investment cost for a cooling capacity of 40 kW with storage, is between 2500 and 5000 €/kW_{cool} [139,140]. The range of costs of FPC is 200–500 €/m^2 while the ETC and CPC is between 500 and 800 €/m^2[130]. The cost of PTC is similar to the FPC according the consulted manufacturer sources in this work.

The parabolic-trough collectors generally require more supervision and maintenance since the reflectors must be cleaned and checked for leaks two to four times per year [62] Table 6.

5. Results and discussion

Fig. 7 shows a quantitative comparison between the annual efficiency of the different solar collectors considered. For a 1E-chiller the ETC collector is the only with better performance (due to the low operating temperature and better use of diffuse radiation), followed by PTC collectors, this increased performance is not accompanied of higher energy production because the annual energy received $G_{t,\ annual}$ by the ETC is lower. For the 2E-chiller PTC collector are more efficient as expected due to the better performance to temperatures near 150 °C.

Figs. 8 and 9 show the results. The mean annual solar fraction, f, is on the ordinate axis and the specific collector area, A_{spec}, is on the abscissa, storage at this case was not considered.

In the collector comparison study, curves are observed to be grouped for location in all cases, in such a way that the PTC has the highest f, while it is intermediate for the CPC and ETC and the lowest for the FPC. The differences between groups of curves are wider with the 2E-chiller and only the PTC remains at the like f whether connected to 1E or 2E chiller.

In Fig. 9, for the hotel in Madrid, the f curves show a qualitative behaviour similar to the office-building in the same location (Fig. 8), but decrease significantly because the cooling load per m^2, E_{cool}, for the hotel does not match the solar radiation (Fig. 5). The PTC curves are the highest f ($f > 40\%$ in Madrid and $f > 30\%$ in Copenhagen) with both types of absorption chillers, while they are lower for the rest of

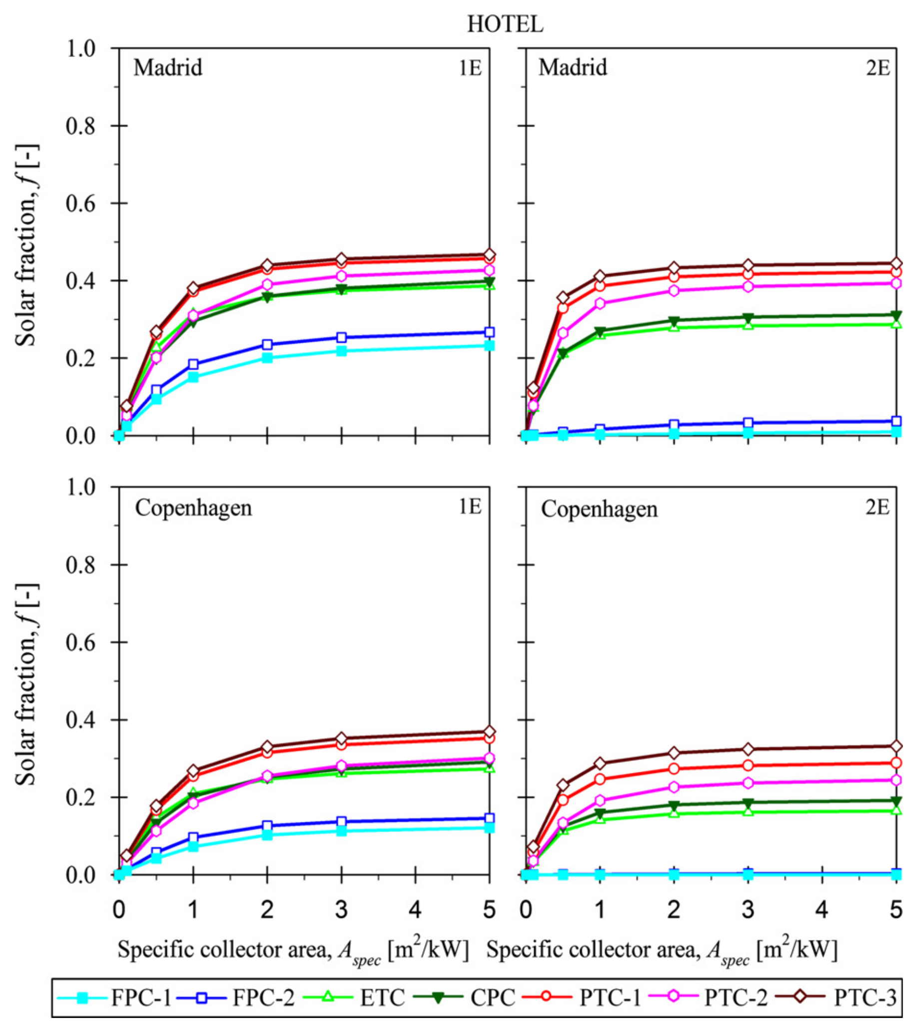

Fig. 9. Collector comparison of annual solar fraction, for the hotel in Madrid and Copenhagen, connected to single-effect (1E) and double-effect (2E) absorption chillers [V_{sto}/A_c=0 m].

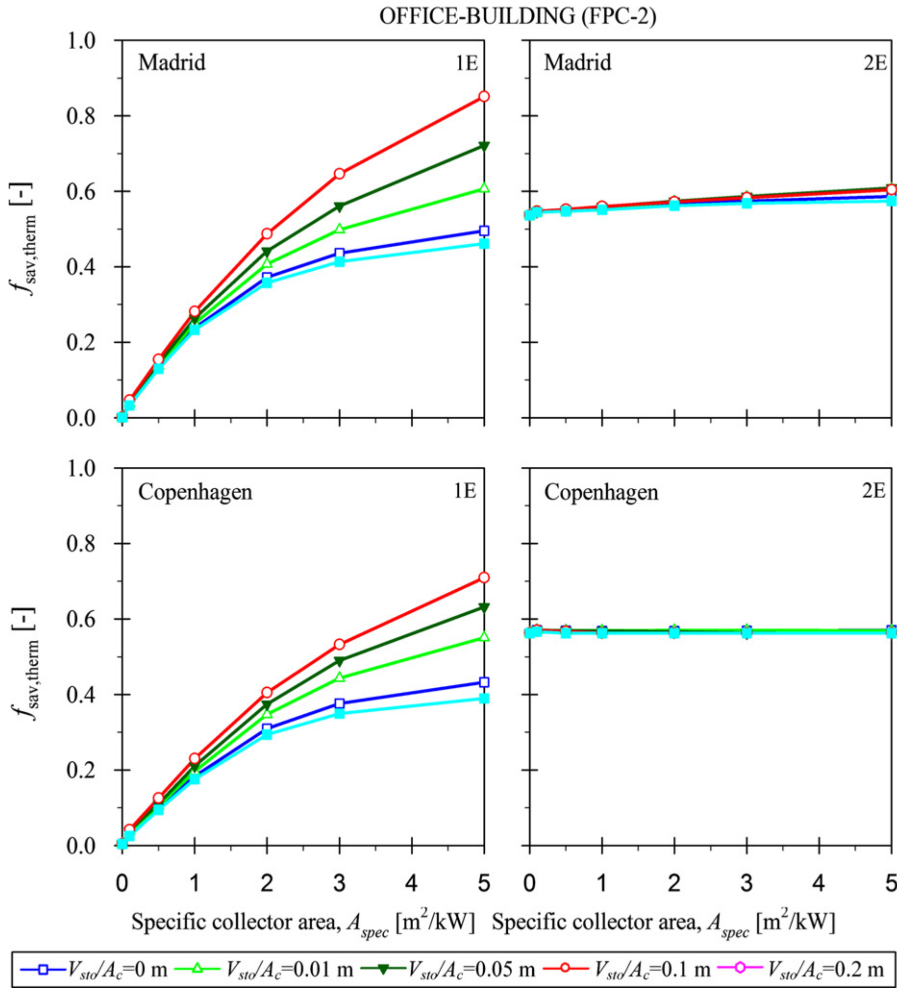

Fig. 10. Comparison of annual fractional thermal energy savings, $f_{sav,therm}$, for the flat plate collector (FPC-2) connected to single-effect (1E) and double-effect (2E) absorption chillers installed on the office-building in Madrid and Copenhagen, for different storage capacity, V_{sto}/A_c. [Madrid-E_{ref}=36,600 kW h and Copenhagen-E_{ref}=17,300 kW h, tank heat losses not considered].

the collectors ($f<$ 30–40%). This could be improved by incorporating a thermal storage system in the facility.

The lowest f in the whole study are recorded for the hotel in Copenhagen (Fig. 9), due to the combined effect of an increased cooling load because of the type of building with an unfavorable profile and low thermal energy supplied by the collectors due to environmental conditions. So the solar air-conditioning system is not practical, unless it is provided with thermal storage. In this case, the only collectors with an f that could be considered higher ($f>$ 30%) are the PTC.

Regarding the thermal storage, solar fraction must be obviously influenced because storing the energy produced by the collectors may be available in the system at a later time provided heat losses in storage are minimized. According to the calculations the $f_{sav,therm}$ is greater the higher the storage for the FPC-2 in the office building with 1E-chiller (Fig. 10), $f_{sav,therm}$ do not increase when area collector with the 2E-chiller, due to the poor efficiency of the FPC to the driving temperature of the 2E-chiller (Fig. 7).

The annual fractional of thermal energy savings for the PTC-1, in office-building (Fig. 11), increases with the storage volume, for the 1E and 2E chillers case. The 2E-chiller produces maximum heat energy savings around of 55% compared to the 1E-chiller for Madrid and Copenhagen (A_{spec}=0 m^2/kW) (Figs. 10 and 11). Other studies have also demonstrated thermal energy savings produced by the assisted solar cooling connected to 1E and 2E chillers [47].

Fig. 12 shows that the collector that has a lower cost of energy production is the PTC-2 according to its investment cost. The $LCOE_{cool}$ is similar for PTC and FPC with 1E-chiller, depending on the selected collector $\Delta_{LCOE}=-1\%$ (PTC-1 < FPC-1) and $\Delta_{LCOE}=0\%$ (PTC-3 ≈ FPC-2). This comparison cannot be performed with 2E-chiller because the FPC does not reach the necessary driving temperature for a 2E chiller and therefore it has a low solar fraction to be considered. The PTC have lower cost of energy production that ETC ($\Delta_{LCOE}=-23\%$) and CPC ($\Delta_{LCOE}=-17\%$). The solar cooling system with 2E-chiller has a similar $LCOE_{cool}$ than 1E-chiller system despite higher auxiliary thermal energy savings of 2E-chiller. These results however present a high sensitivity to the capital cost of the chillers. The lower $LCOE_{cool}$ is in Madrid due to the higher available solar radiation and in Hotel due to a greater demand of refrigeration for the similar investment and operation and maintenance cost. The solar cooling system can be more competitive by increasing the cooling loads for the same chiller cooling capacity, since it increases the amortization of the capital investment costs and thus originates a lower $LCOE_{cool}$.

6. Conclusions

In this work, it has been done a comprehensive literature survey on worldwide air-conditioning and refrigeration facilities fed by a PTC solar field. Also, it has been given extensive references on new

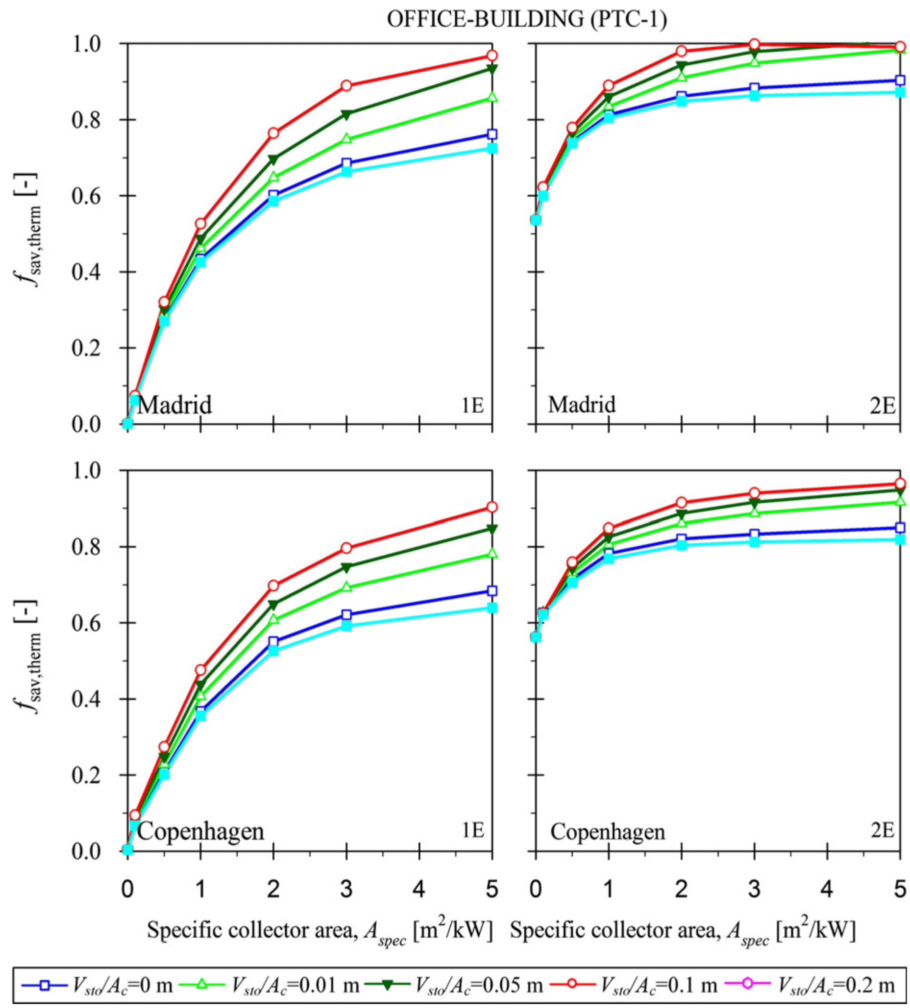

Fig. 11. Comparison of annual fractional thermal energy savings, $f_{sav,therm}$, for the parabolic-trough collector (PTC-1) connected to single-effect (1E) and double-effect (2E) absorption chillers installed on the office-building in Madrid and Copenhagen, for different storage capacity, V_{sto}/A_c. [Madrid-E_{ref}=36,600 kWh and Copenhagen-E_{ref}=17,300 kWh, tank heat losses not considered].

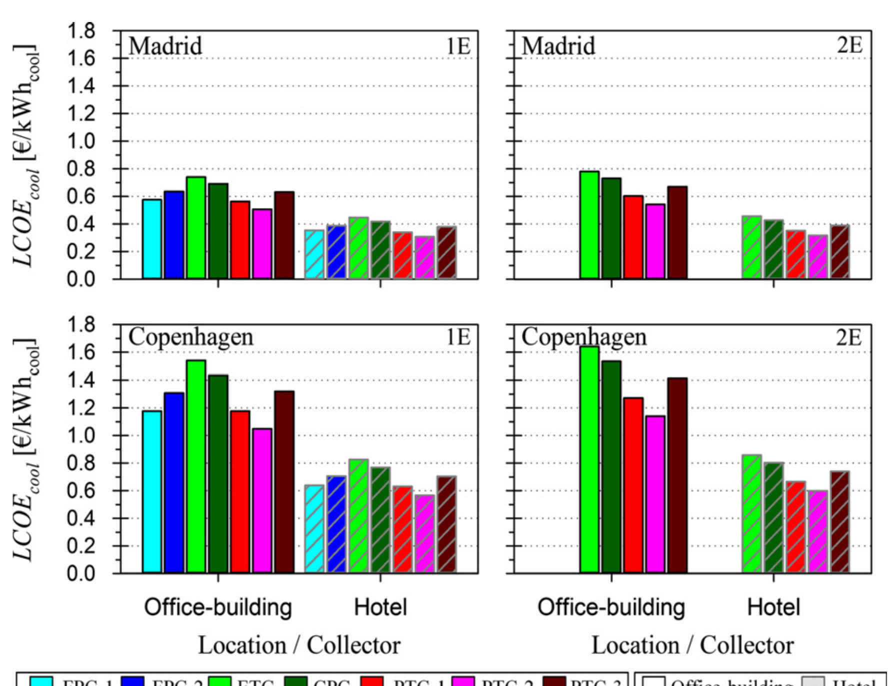

Fig. 12. Levelized cost of energy cooling, $LCOE_{cool}$, for the office-building and the hotel in Madrid and Copenhagen, connected to a single-effect (1E) and double-effect (2E) absorption chillers [A_{spec}=3 m^2/kW, V_{sto}/A_c=0 m, collectors with $f<0.1$ are not considered].

developments in PTC suitable to be used in process heat and double effect absorption chillers. Findings of this study demonstrate that the yearly rate of grow of this type of installations, excluding 2010 year, is still low, about 4 installations per year especially if compared to the rates of installations fed by FPC or ETC. However, according to the market potential, it is expected an increased rate once demonstrated the availability of small PTC. In regard to the new PTCs, it must be highlighted that in spite of the existence of a growing commercial offer, in most of the cases, testing data and standardization are still scarce.

The performance of PTC in air conditioning applications has been compared to other solar thermal collectors on the basis of the SACE methodology, enlarging its analysis capabilities by adding dynamic simulation estimations of the parameters and incorporating the parabolic trough collector for solar refrigeration and air-conditioning applications.

In all the analysed cases, the PTC solar fraction is higher than for the rest of the collectors, the ETC and CPC have low intermediate fractions, and the FPC have the lowest. The results obtained suggest that for the considered cases, solar cooling installations should be designed with a specific area of collector range of 0–3 m^2/kW for single-effect (1E) absorption chillers and 0–2 m^2/kW for double-effect (2E) absorption chillers, as there is only a slight increase in the solar fraction above those ratios.

The PTC has a similar solar fraction when connected to a single-effect or double-effect absorption chiller, while the rest of the collectors have lower solar fractions with 2E-chillers, for the same specific area collectors. The annual solar fraction in the hotel is lower than the office, without a thermal storage system, due to the mismatch between the solar radiation and the cooling load. The solar fractions in Copenhagen are all lower than in Madrid, because both the available solar radiation and ambient temperatures are lower. If a thermal storage tank is included in the solar system, the required auxiliary heating can be also lowered provided heat losses are minimized. Finally, PTC present a similar levelized cost of energy for cooling that FPC and lower that ETC and CPC.

Nomenclature

a_1	heat loss coefficient [W/(m^2 K)]
a_2	temperature dependence of the heat loss coefficient [W/(m^2 K^2)]
A_{building}	building area [m^2]
A_c	collector area [m^2]
A_{spec}	specific collector area [m^2/kW]
b_{1L}, b_{2L}	longitudinal direct incident angle modifier coefficients [-]
b_{1T}, b_{2T}	transversal direct incident angle modifier coefficients [-]
C	concentration ratio of collector [-]
COP_{rated}	rated coefficient of performance [-].
crf	annual amortization factor [-]
E_{aux}	annual auxiliary energy required [kW h]
E_c	thermal energy given by the solar system per m^2 collector [W h/m^2]
E_{cool}	cooling load per m^2 building [W h$_{\text{cool}}$/m^2]
E_{hot}	thermal energy demanded by the air-conditioning system per m^2 building [W h/m^2]
E_{ref}	annual reference, non-solar heating system, auxiliary energy required [kW h]
f	annual solar fraction [-]
f_h	hourly solar fraction [-]
$f_{\text{sav,therm}}$	annual fractional thermal energy savings [-]
$G_{b,t}$	direct solar radiation on a surface tilted [W h/m^2]
$G_{d,h}$	diffuse solar radiation on a horizontal surface [W h/m^2]
$G_{d,t}$	diffuse solar radiation on a surface tilted [W h/m^2]
G_h	global solar radiation on a horizontal surface [W h/m^2]
G_t	global solar radiation on a surface tilted [W h/m^2]
k_d	is the interest rate debt [-]
K_{fuel}	annual fuel cost [€/year]
$k_{\text{insurance}}$	annual insurance rate [-]
K_{invest}	capital cost of the total investment of the plant [€]
$K_{O\&M}$	annual cost of operation and maintenance [€/year]
$K_{\theta b}$	direct incident angle modifier [-]
$K_{\theta d}$	diffuse incident angle modifier [-]
$K_{\theta L}$	longitudinal direct modifiers of incident angle [-]
$K_{\theta T}$	transversal direct modifiers of incident angle [-]
$LCOE_{\text{cool}}$	levelized cost of energy cooling [€/kW h]
n	project time life [years]
$\dot{Q}_{\text{cool}}$	cooling capacity [kW]
T_a	ambient temperature [K]
T_{hot}	driving temperature of chiller [°C]
T_m	mean temperature of the working fluid [K]
V_{sto}	storage tank volume [m^3]
Δ_{LCOE}	increment of levelized cost of energy cooling [%]
η	overall efficiency of collectors [-]
η_0	zero-loss collector efficiency [-]
θ	angle of incidence [°]

Acronyms

1E	Single-effect
2E	Double-effect
3E	Triple-effect
COP	Coefficient of performance
CPC	Compound parabolic concentrator
CSP	Concentrating solar power
DHW	Domestic hot water
DSG	Direct steam generation
ETC	Evacuated tube collector
FPC	Flat-plate collector
HX	Heat exchanger
IEA	International energy agency
n/a	Not available information
PTC	Parabolic-trough collector
PLFR	Parabolic linear Fresnel reflectors
PVT	Photovoltaic and thermal
SACE	Solar air-conditioning in Europe
SC	Solar cooling
SCP	Solar cooling and power
SK	Solar Keymark certification
SEGS	Solar electric generating systems
SHC	Solar heating and cooling
SHCP	Solar heating, cooling and power
SHX	Solution heat exchanger
SRCC	Solar rating and certification corporation

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